

Manufacturing and testing of a gamma type Stirling engine

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Abstract

In this study, a gamma type Stirling engine with 276 cc swept volume was designed and manufactured. The engine was tested with air and helium by using an electrical furnace as heat source. Working characteristics of the engine were obtained within the range of heat source temperature 700–1000 °C and range of charge pressure 1–4.5 bar. Maximum power output was obtained with helium at 1000 °C heat source temperature and 4 bar charge pressure as 128.3 W. The maximum torque was obtained as 2 N m at 1000 °C heat source temperature and 4 bar helium charge pressure. Results were found to be encouraging to initiate a Stirling engine project for 1 kW power output.

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1. Introduction

The rapid depletion of our natural resources has focused attention on new energy sources and on effective means of energy conservation. One of the means of effective energy conservation is the Stirling engine [1,2]. The Stirling heat engine was first patented in 1816 by Robert Stirling. Since then, several Stirling engines based on his invention have been built in many forms and sizes [3]. The Stirling engine is an externally heated, environmentally very clean engine having high theoretical cycle efficiency. The engine works with a closed cycle and uses several

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gases as working fluid; among them, helium and hydrogen give higher performance. To energize the engine, many energy sources, such as, combustible materials, solar radiation, geothermal hot water, radioisotope energy and so on, can be used. As mechanical arrangement, many simple models of Stirling engine have been designed and constructed which consist of about 20 parts [3,4].

In solar modules, Stirling-Dish, the solar radiation is converted to electricity in three stages. In the first stage, radiation is converted to heat by focusing the solar radiation onto a light absorbing heat pipe by means of a parabolic reflector. In the second stage, the heat is converted to mechanical power by a Stirling engine. In the final stage, the mechanical power is converted to electricity by an alternator. Qualities of the energy conversion from radiation to heat, from heat to mechanical power and from mechanical power to electricity are defined as collector efficiency, thermal efficiency and alternator efficiency, respectively. The efficiency of the complete system is defined as overall efficiency or from light to electricity efficiency.

In solar energy investigations, the first use of Stirling engine was made by Parker and Malik in the 1960s [1]. Their experimental facility consisted of a Fresnel lens, a light absorber and a Stirling engine. The eutectic mixture of potassium and sodium convected the heat from the absorber to the heater head by means of evaporation in the absorber and condensation on the heater head.

In a solar energy project undertaken by Shaltens et al. [5], sponsored by the US Department of Energy and organized by NASA Lewis Research Center, two different free pistons Stirling engines were tested. In the experimental facility, a sodium heat pipe receiver was used. The engine manufactured by Cummins Motor Company provided efficiencies reaching 32% from light to electricity. The engine manufactured by Stirling Technology Company provided 30.7% efficiency from light to electricity.

Cummins Power Generation Inc. manufactured and tested two free piston Stirling engines [6]. Engines were tested on a plant made of stretch membrane reflectors. The first engine was designed for 4 kW electric power output and test results were encouraging. The second engine was designed for 5 kW electric power output, and gave 6 kW electric power output at 675 °C heater head temperature. The second engine provided 30% light to electric efficiency.

Gupta et al. [7] developed 1 and 1.9 kW solar-powered Stirling engines for rural applications. Engine efficiencies obtained were between 5.5% and 5.7% and overall efficiency obtained was 2.02%.

In an experimental study, carried out by Hoshino et al. [8] and promoted by the National Space Development Agency of Japan, a semi free piston Stirling engine was tested. From light to electric efficiency of the test facility was measured as 20%. The thermal efficiency of the experimental facility was determined as 33%.

Prodesser [9] developed a Stirling engine, heated by the flue gas of a biomass furnace, for electricity production in rural villages. With a working gas pressure of 33 bars at 600 rpm and a shaft power of 3.2 kW, an overall efficiency of 25% was obtained.

At present, vehicular applications of Stirling engines are prevented by their large volume and weight. However, due to their silent operation and harmless burn out

gases Stirling engines could be the power source of passenger cars in the 21st century [10–12].

Since the conversion efficiency of Stirling based electric generation is three times higher than that of radioisotope thermoelectric generation, a general purpose Stirling engine project has been started by the US Department of Energy and NASA for use in space missions [13].

The present study is concerned with the design and manufacture of a double piston unique displacer gamma type Stirling engine with hot end connection. The advantage of using two pistons is the augmentation of heat transfer area and supply of more heat to the working gas at expansion period. In gamma type Stirling engines, either the hot end or the cold end of displacer cylinder can be connected to the expansion cylinder. Hot end connection serves the advantage of larger cyclic work [14].

2. Test engine

The schematic illustration of the test engine and its specifications are shown in Fig. 1 and Table 1.

The crankcase consisted of a body and two side lids. The body was processed from a piece of circular steel duct. To prevent the escape of working fluid, the

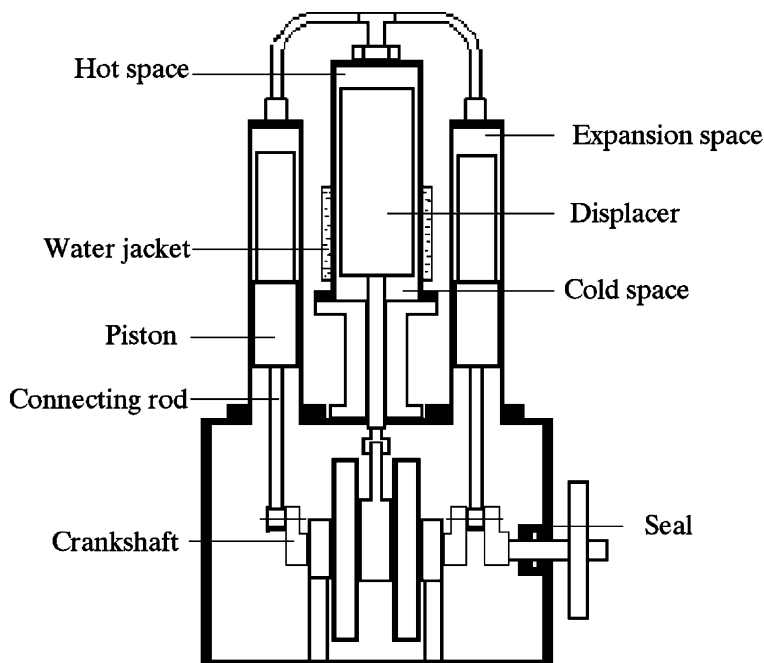


Fig. 1. Schematic illustration of the test engine.

Table 1

Technical specifications of the test engine

Engine type	γ
Swept volume	276 cc
Phase angle	90°
Working fluid	Helium and air
Cooling system	Water cooled
Compression ratio	1.82
Maximum engine power	128.3 W at 891 rpm
Engine speed	1589 rpm
Dead volume	169 cc

crankshaft was totally confined in the crankcase. The motion of crankshaft was taken out by means of a second shaft embedded on the side lid of the crankcase. The embedding of the second shaft on the side lid was made by means of a leak free bearing. The expansion cylinders were located on two sides of the displacer cylinder symmetrically and all three were connected to each other by hot ends.

The sections of crankshaft were manufactured from different steels having appropriate properties and then joined to each other by hard fitting and welding. Both ends of the crankshaft were embedded in ball bearings. On the crankshaft, there were three connecting rod pins corresponding to piston and displacer rods. There was a 90° phase angle between the piston and displacer pins. To keep the vibration lower, two balance drums were imposed on the crankshaft.

Pistons consisted of a body and a hub. The body of the pistons was made of spheroid-graphite cast iron because of its low friction, high impact resistance and machining simplicity. The contact surface of the pistons to the surface of cylinder liner was machined to super-finish quality by grinding and polishing. For the protection of pistons and cylinder liners from high temperature, pistons were extended with hubs. Because of their resistance to oxidation under high temperature, hubs were made of ASTM duct. The pistons were connected to the crankshaft by light rods made of aluminum.

The cylinder liners were made of hardened AISI 4080 steel. The inner surfaces of the liners were finished by honing. Pistons and liners were adapted to each other with 0.02 mm working clearance and no seals were used on the pistons. To minimize the leakage of working fluid through the clearance between piston and liners, the length of pistons was twice that of the diameter. The heads of cylinders, which are exposed to high temperature, were made of ASTM duct.

The displacer and its cylinder, which works as heater head, were also made of ASTM steel duct. Two different displacer cylinders were manufactured and tested. One had a smooth inner surface and the other had an augmented inner surface with triangular spanwise slots. Between the displacer and its cylinder, 1 mm gap was left for the flow of working fluid. The external surface of the lower half of the displacer cylinder was cooled by housing in a cooling water jacket. The rod of displacer was made of hardened AISI 4080 steel, which works in a cast iron bed.

The stroke volume of displacer and pistons was equal to each other, 138 cc. The total dead volume of the engine was 169 cc. The compression ratio was measured as 1.82. The surface area of the displacer and inner surface area of the displacer cylinder were 425 and 505 cm², respectively. The inner surface of expansion cylinders was 170 cm². The inner surface area of the connecting pipe was about 100 cm². The total heat transfer area of the engine was approximately 1200 cm².

3. Experimental facilities and testing procedure

A heat source, an electrical furnace, NUVE, was used after some adaptations. The furnace was fitted with a thermometer, NEL, with 1 °C of accuracy. The furnace was adjustable to any temperature between 0 and 1500 °C. The hot ends of the displacer and expansion cylinders were inserted into the furnace through grooved openings on the wall of furnace. Loading of the engine was done by a dynamometer manufactured in the laboratory to measure the moments lower than 2.5 N m and has 0.003 N m of accuracy. The speed of the engine was measured by a digital tachometer, METEK, of 0–10,000 rpm measurement range and 1 rpm accuracy. For the regulation of working fluid pressure, a pressure-reducing valve was used. The charge pressure was measured with a Bourdon tube pressure gauge, SEDEF, 0.1 bar accuracy and 0–10 bar measurement range. Ambient air and helium were used as working fluid. The lower half of the displacer cylinder was kept at 20 °C by circulating tap water through the water jacket around it. The schematic illustration of the test rig is seen in Fig. 2.

During development processes, the engine was run many times under different working conditions. After ensuring that no more mechanical and thermal problems

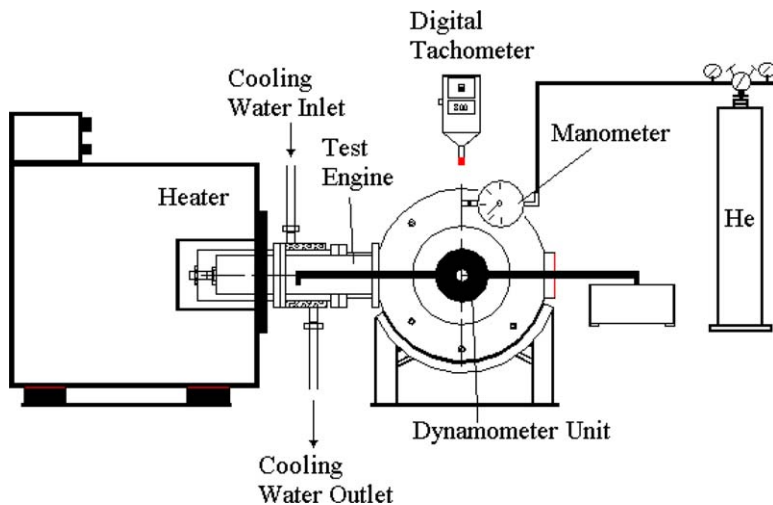


Fig. 2. Schematic illustration of the test rig.

remained, the systematic tests were conducted to obtain the working characteristics of the engine. At the heat source temperatures below 700 °C, the application of charge pressure reduced the performance of the engine. Therefore, systematic testing was conducted for furnace temperatures above 700 °C.

As the furnace temperature was constant at any desired value, the speed of the engine was measured versus the torque, applying different charge pressures. The charge pressure was increased by 0.5 bar at each step. At a certain value of charge pressure, the performance of the engine reached a maximum. After this, increasing the charge pressure caused a gradual decrease in the engine performance and finally, the engine tended to stop. Then, the temperature of the heat source was increased by a 100 °C. The systematic testing of the engine was completed within the aimed limits of the charge pressure and heat source temperature by repeating the same operations at subsequent temperature stages.

4. Experimental results and discussion

Results obtained at various temperatures of heat source using air and helium are shown in Fig. 3. It is seen that an increase in the heater temperature results in an increase in power output. Comparison of curves given for helium and air at the same conditions shows that for helium the engine power output is about twice that of air. For example, at 800 °C heat source temperature and 3 bar charge pressure, helium gives 44.06 W of power while air gives 24.5 W. The curves indicate that the increase of power with heat source temperature is limitless. Therefore, progress made in material technology, resistance to high temperature, will improve the performance of Stirling engines.

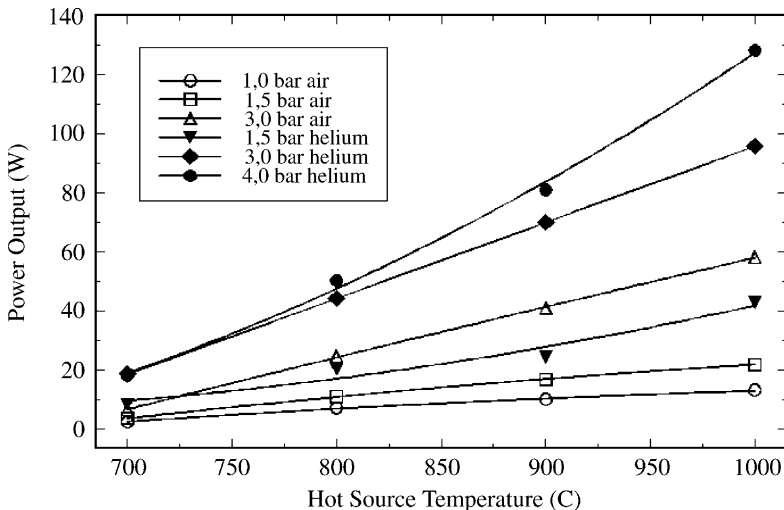


Fig. 3. Variation of brake power with heat source temperature.

Fig. 4 shows the variation of power output as a function of charge pressure for various heat source temperatures and different working fluids. An increase in the power output is shown to depend on the charge pressure at a certain level.

As seen in Fig. 4, optimum charge pressure depends on the working fluid and heat source temperature. For the same heat source temperature, helium has a higher optimum charge pressure. Higher thermal conductivity of helium compared to air improves the heat transfer between working fluid and solid walls. As the heat source temperature increases, the optimum charge pressure also increases. Maximum engine power is obtained at 1000 °C as 128.3 W for helium and 58 W for air under 4 bar helium and 3.0 bar air charge pressure, respectively. Increasing the charge pressure over the optimum value causes not only decreasing of power, but also increases the vibration of the engine. It was also noted that, when the charge pressure was at the ambient level, the engine started race about 500 °C heat source temperature. But at the higher charge pressures, the engine started race at higher furnace temperatures.

Fig. 5 shows the variation of output power with engine speed at 1000 °C heat source temperature and different rates of helium and air charge pressures. Since the power is a function of speed and torque, the higher the speed and torque the higher the output power. Because of inadequate heat transfer rate, a decrease in power output is observed after a certain engine speed.

Fig. 6 shows the variation of torque with engine speed at 900 and 1000 °C heat source temperatures, and 3 bar charge pressure. In the same figure, the torque–speed characteristics of air and helium are compared. When air is used, the load free speed of the engine is less than 800 rpm. When helium is used, the load free speed of the engine exceeds 1400 rpm. While the load increases gradually, the speed lessens gradually as well. However, after a certain value of the load, engine speed

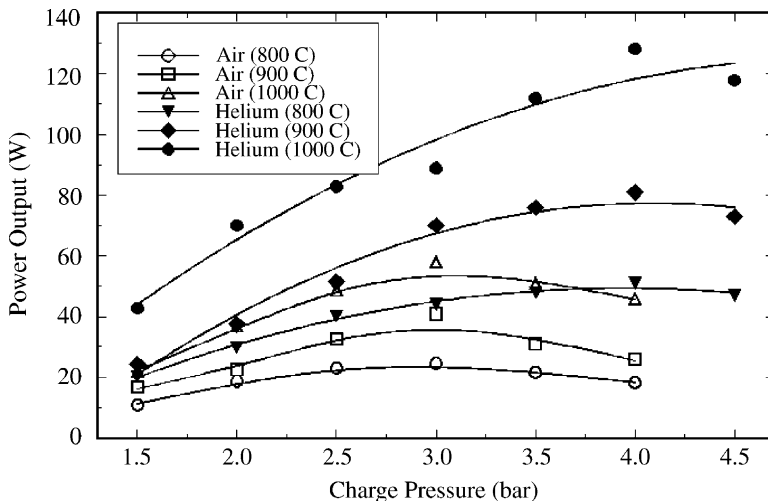


Fig. 4. Variation of brake power with charge pressure.

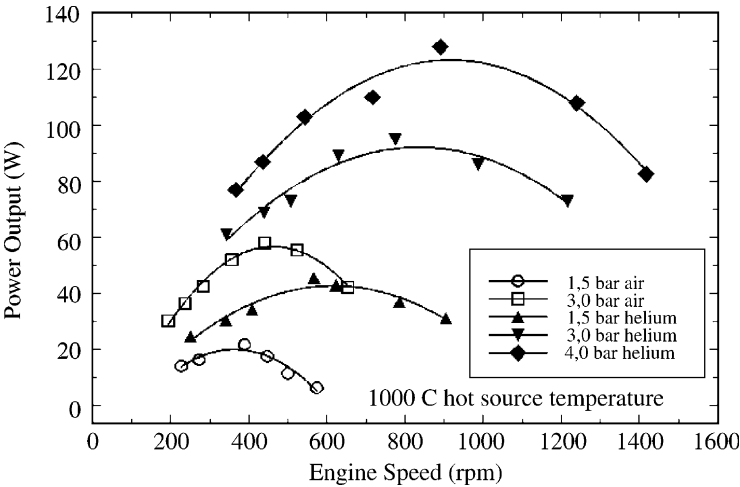


Fig. 5. Variation of brake power with engine speed.

decreases to an irregular range and further increase in load causes the engine to stop. The irregularity in speed at high load is caused by the insufficient kinetic energy stored by the flywheel to compress the working fluid. During the compression process, due to the energy transfer from the flywheel to the working fluid, the speed decreases. In the subsequent expansion process, the speed increases again. Due to such variation in the speed, the torque becomes too difficult to measure.

The effect of augmentation of heat transfer surface is illustrated in Fig. 7. When the heat transfer surface of the engine is increased about 20%, the maximum power

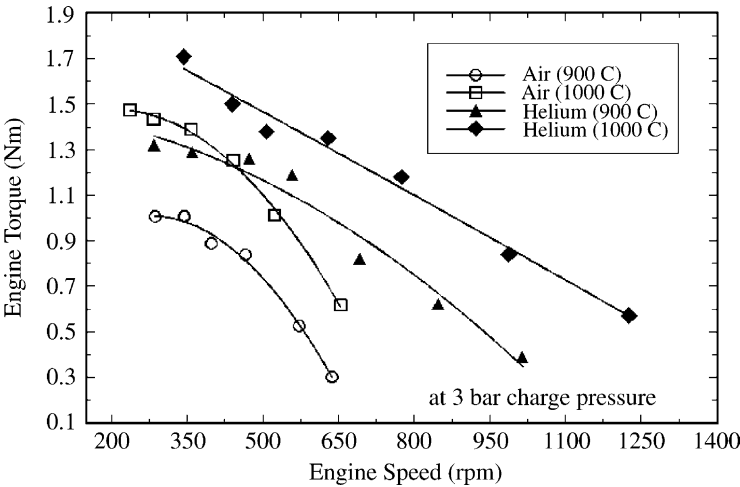


Fig. 6. Variation of engine torque with engine speed.

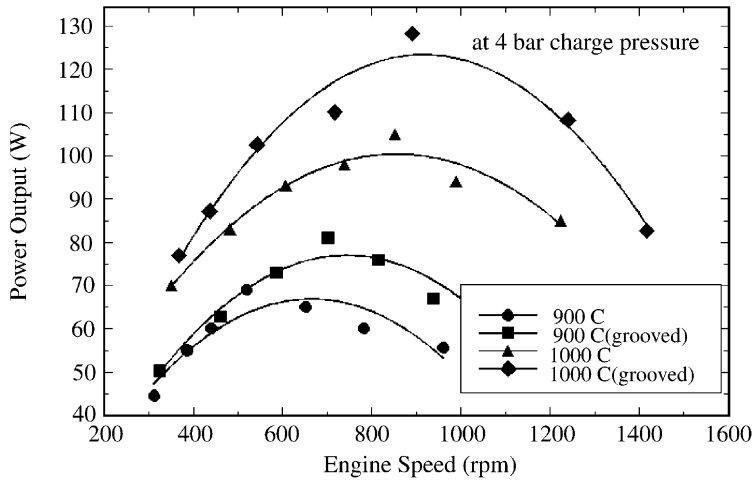


Fig. 7. Effect of augmentation of heat transfer area on engine power.

increases from 104 to 128 W, while a 20% increase in heat transfer area, the increase of power is 23%. When the heat transfer area of the displacer cylinder is augmented by grooving slots, a slight change in compression ratio occurs.

When the engine is stopped and restarted after a few seconds later, a transient running period is observed. In transient period, the engine provides about 30% higher power than the steady state. From this, it is understood that during the stopping period the temperature of the heater head of the engine reaches a higher value and when the engine is restarted, a better heat transfer occurs between the working fluid and heat transfer surfaces due to higher temperature difference.

Typically, the heater head temperature of the engine is chosen as a fundamental parameter of the experimental study. In this study, however, due to the difficulty of measuring the temperature of the heater head, the furnace temperature was chosen as the fundamental parameter. The 30% power difference between steady state and transient working is an indication of the difference between the furnace temperature and the heater head temperature.

5. Conclusion

The prototype provided 0.464 W power per cc swept volume and 0.107 W power per cm^2 heat transfer area. The use of helium more than doubled the power output of the engine. At each heater head temperature a different optimum charge pressure was observed. The augmentation of displacer cylinder surface provided a proportional increase in power. The difference between furnace temperature and engine heater head temperature was found to be not significant.

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